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Evaluation of the EHL Film Thickness and Extreme Pressure Additives on Gear Surface Fatigue Life

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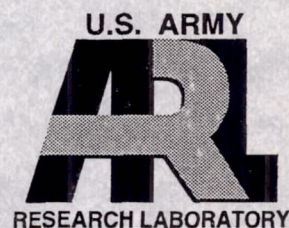
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EVALUATION OF THE EHL FILM THICKNESS AND EXTREME PRESSURE ADDITIVES ON GEAR SURFACE FATIGUE LIFE

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Abstract

Surface pitting fatigue life tests were conducted with seven lubricants, using AISI 9310 spur gears. The test lubricants can be classified as synthetic polyol-esters with various viscosity's and additive packages. The lubricant with a viscosity that provided a specific film thickness greater than one and with an additive package produced gear surface fatigue lives that were 8.6 times that for lubricants with a viscosity that provided specific film thickness less than one. Lubricants with the same viscosity and similar additive packages gave equivalent gear surface fatigue lives.

Introduction

Aircraft transmission for helicopters, turboprop, and geared turbo-fan aircraft are required to have high reliability and provide several thousand hours of operation between overhaul. In addition, they should be lightweight and have very high efficiency to minimize operating costs for the aircraft.

Most of the aircraft operating today are using turbine engine lubricants to lubricate the transmissions. While these lubricants provide good lubrication, thermal stability, and low operation temperatures for the turbine engines they are less than optimum for good reliability and long life for transmissions.

Tests with rolling element bearings have shown that bearing life is affected by the lubricant elastohydrodynamic (EHL) film thickness (refs. 1 and 2). When the EHL film thickness divided by the composite surface roughness h/σ is less than 1 the life of rolling element components is considerably reduced.

In gearing the effect of operating with an h/σ of less than one is more pronounced than it is with bearings. This is because of the higher sliding conditions encountered with gearing which causes increased surface heating and higher friction coefficients resulting in reduced EHL film thickness, surface fatigue life, and increased wear or scoring risk.

Gear tests conducted with several lubricant additives have shown that the gear surface fatigue life can be improved somewhat with the right choice of additives (refs. 3 and 4). Lubricants with the same viscosity but with different additives produced gear surface fatigue lives with a difference of five to one. The above mentioned tests indicted the necessity of having the proper additive in the lubricant but did not determine what affect different lubricant viscosity of the same base stock would have on gear fatigue life.

The effect of the EHL film thickness on scoring and wear under various slide to roll ratios was determined in reference 5 using rolling sliding cylinders. When the specific film thickness, Λ or h/σ , was less than or equal to

0.3 the rolling sliding cylinders experienced wear and scoring, and indicated an increase in friction coefficient and temperature. These tests (ref. 5) also showed an increase in scoring load capacity with EP additives in the lubricant.

Lubricant suppliers have recognized the need for supplying better lubricants for the modern gearboxes that operate at increased power density (ref. 6). Tests have shown that lubricants with the proper base stock, viscosity, and additives can improve the load capacity and efficiency of transmissions.

The research work reported herein was undertaken to investigate the effects of lubricants with the same base stock but with different viscosity's on the surface fatigue life of AISI 9310 spur gears. The objectives were: (1) to investigate the effect of seven different lubricants on the surface fatigue life of hardened steel spur gears, (2) to compare the gear fatigue life with six of the seven lubricants to a reference lubricant, and (3) to determine the effects of lubricant viscosity and specific EHL film thickness on the surface fatigue life of carburized and hardened spur gears.

To accomplish these objectives, one lot of spur gears was manufactured from a single heat of (CVM) AISI 9310 material. The test gears were case carburized, hardened and ground to the same specifications. The gear pitch diameter was 8.89 cm (3.5 in.). The lot of gears was divided into seven groups, each of which was tested with a different lubricant. All of the test lubricants can be classified as synthetic polyolester with different viscosity properties and additives. Test conditions include a bulk gear temperature of 350 K (170 °F), a pitch line maximum Hertz stress of 1.71 GPa (248 ksi) and a speed of 10 000 rpm.

Apparatus and Procedure

Gear Test Apparatus

The gear surface fatigue tests were performed in the NASA Lewis Research Center's gear test apparatus (fig. 1(a)). This test rig used the four-square principle of applying the test gear load so that the input drive only needs to overcome the frictional losses in the system.

A schematic of the test rig is shown in figure 1(b). Oil pressure and leakage flow are supplied to the load vanes through a shaft seal. As the oil pressure is increased on the load vanes inside the slave gear, the loop torque is applied. This torque is transmitted through the test gears back to the slave gear, where an equal but opposite torque is maintained by the oil pressure. This torque on the test gears, which depends on the hydraulic pressure applied to the load vanes, loads the gear teeth to the desired contact or Hertz stress level. The two identical test gears can be started under no load, and the load can be applied gradually, without changing the running track on the gear teeth.

Separate lubrication systems are provided for the test gears and the main gearbox. The two lubrication systems are separated at the gearbox shafts by pressurized labyrinth seals. Nitrogen is the seal gas. The test gear lubricant is filtered through a 5- μ m-nominal fiberglass filter. The test lubricant can be heated electrically with an immersion heater. The temperature of the heater is controlled to prevent overheating the test lubricant. A water cooler and temperature controller is also provided in the test oil and slave oil system to control the inlet oil temperature.

A vibration transducer mounted on the gearbox is used to automatically shut off the test rig when a gear surface fatigue or tooth fracture occurs. The gearbox is also

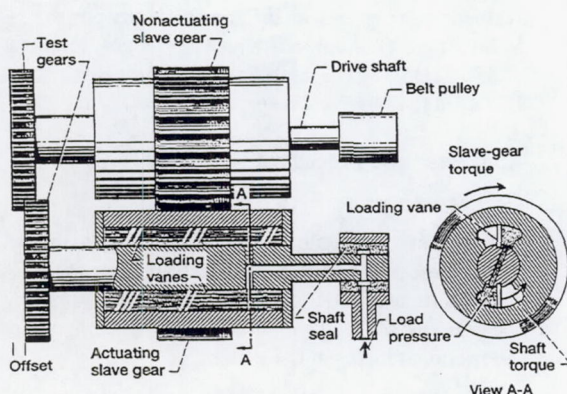
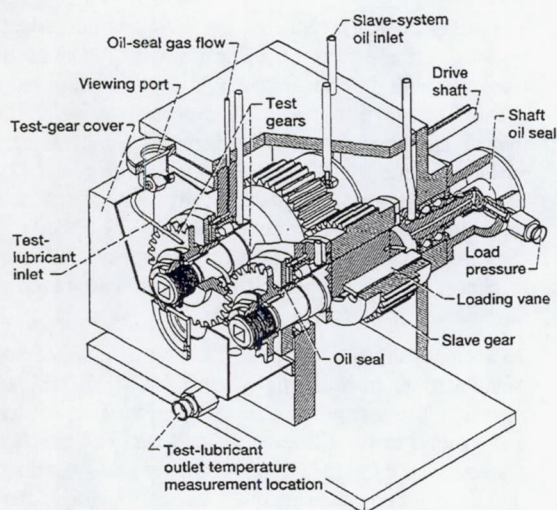


Figure 1.—NASA Lewis Research Center's gear fatigue test apparatus.

automatically shut off if there is a loss of oil flow to either the main gearbox or the test gears, if the test gear oil overheats, or if there is a loss of seal gas pressurization.

The belt-driven test rig can be operated at several fixed speeds by changing pulleys. The operating speed for the test reported herein was 10 000 rpm. The four test rigs are operated 24 hr a day, 7 days per week to provide the large number of test cycles that are required for surface fatigue testing.

Test Gears

Dimensions for the test gears are given in Table 3. The gear pitch diameter was 8.89 cm (3.5 in.). All gears have a nominal surface finish on the tooth face of 0.406 μ m (16 μ in.), rms, and a standard 20° involute profile with tip relief. Tip relief was 0.0013 cm (0.0005 in.), starting at the highest point of single tooth contact.

The test gears were manufactured from consumable-electrode vacuum-melted (CVM) AISI 9310 steel from the same heat of material. The nominal chemical composition of the material is given in Table 1. All sets of gears were case carburized and heat treated in accordance with the heat treatment schedule of Table 2. Figure 2 is a photomicrograph of an etched and polished gear tooth showing the case and core microstructure of the AISI 9310 material. This material has a case hardness of Rockwell C60 and a case depth of 0.97 mm (0.038 in.). The nominal core hardness was Rockwell C38.

Test Lubricant

Seven lubricants were selected for surface fatigue endurance tests with the CVM AISI 9310 steel gear test specimens. Lubricant A is an unformulated base stock lubricant with no lubricant additives and with a viscosity in between the MIL-L-7808J and MIL-L-23699 specifications and does not meet either specification. Lubricant A was used as the reference lubricant to compare the results with the other lubricants. Lubricant B is a 5 cSt lubricant meeting the MIL-L-23699 specification.

Table 1.—Nominal Chemical Composition of AISI 9310 Gear Materials

ELEMENT	AISI 9310
Carbon	0.1
Nickel	3.22
Chromium	1.21
Molybdenum	0.12
Copper	0.13
Manganese	0.63
Silicon	0.27
Sulfur	0.005
Phosphorous	0.005
Iron	Bal.

Table 2.—Heat Treatment for AISI 9310 Gears

Step	Process	Temperature		Time, hr
		K	°F	
1	Preheat in air	--	--	--
2	Carburize	1,172	1,650	8
3	Air cool to room temperature	--	--	--
4	Copper plate all over	--	--	--
5	Reheat	922	1,200	2.5
6	Air cool to room temperature	--	--	--
7	Austenitize	1,117	1,550	2.5
8	Oil quench	--	--	--
9	Subzero cool	180	-120	3.5
10	Double temper	450	350	2 each
11	Finish grind	--	--	--
12	Stress relive	450	350	2

Table 3.—Spur Gear Data
[Gear tolerance per AGMA class 12]

Number of teeth	28
Diametral pitch	8
Circular pitch, cm (in.)	0.9975 (0.3297)
Whole depth, cm (in.)	0.762 (0.300)
Addendum, cm (in.)	0.318 (0.125)
Chordal tooth thickness	
(reference), cm (in.)	0.485 (0.191)
Tooth width, cm (in.)	0.635 (0.25)
Pressure angle, deg.	20
Pitch diameter, cm (in.)	8.890 (3.500)
Outside diameter, cm (in.)	9.525 (3.750)
Root fillet, cm (in.)	0.102 to 0.152 (0.04 to 0.06)
Measurement over pins cm (in.)	9.603 to 9.630 (3.7807 to 3.7915)
Pin diameter, cm (in.)	0.549 (0.216)
Backlash reference, cm (in.)	0.0254 (0.010)
Tip relief, cm (in.)	0.001 to 0.0015 (0.0004 to 0.0006)

This lubricant had a small amount of a boundary lubrication additive to provide some boundary lubricating film. Lubricant C meets the MIL-L-7808J specification and had the lowest viscosity of all the lubricants tested and also had a proprietary additive package. Lubricant D was a lubricant developed for helicopter gearboxes under the specification DOD-L-85734 and was also a 5 cSt lubricant with an anti wear additive package. Lubricant E was a 7.5 cSt lubricant with an anti wear additive package meeting a special development specification DERD-2487. Lubricants F and G were 9 cSt ester based lubricants. Lubricant F was a base stock lubricant without additives while lubricant G was a base stock industrial grade lubricant. Six of the seven lubricants could be classified as synthetic polyol-ester base stock lubricants while lubricant E is a polyalkylene-glycol with a small amount of boundary lubrication additive.

The pitch line elastohydrodynamic (EHL) film thickness was calculated by the method of references 8 and 9. The temperature used in the film thickness calculation was taken to be the gear surface temperature at the pitch line which was assumed to be equal the oil outlet temperature even though the temperature of the oil jet lubricating the gear was much lower. It is most probable that the gear surface temperature was higher than the oil outlet temperature based on temperature measurements made in reference 5.

The computed EHL film thickness are given in Table 4 as are initial L ratios (film thickness divided by composite surface roughness, h/σ) at the 1.71 GPa (248 ksi) pitch line maximum Hertz stress.

Test Procedure

After the test gears were cleaned to remove the preservative, they were assembled on the test rig. The 0.635-cm (0.25-in.) wide test gears were run in an offset condition with a 0.30-cm (0.12-in.) tooth-surface overlap to give a load surface on the gear face of 0.28 cm (0.11 in.), thereby allowing for the edge radius of the gear teeth. If both faces of the gears were tested, four fatigue tests could be run for each set of gears. All tests were run in at a pitch-line load of 1225 N/cm (700 lb/in.) for 1 hr, which gave a maximum Hertz stress of 0.756 GPa (111 ksi). The

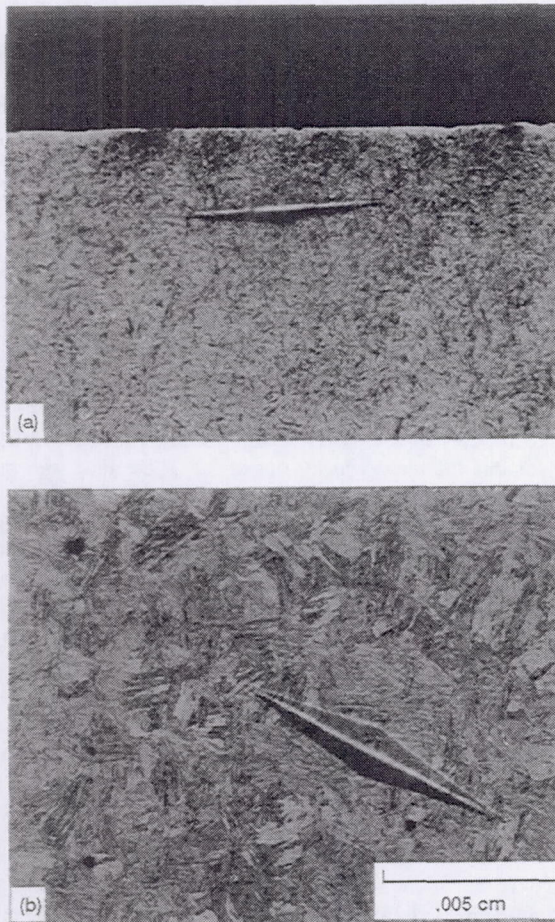


Figure 2.—Photomicrographs of case and core of CVM AISI 9310 spur gears. (a) Case. (b) Core.

Table 4.—Lubricant Properties

NASA identification	Lubricant						
	A	B	C	D	E	F	G
Kinematic viscosity 311 K (100 °F) 372 K (210 °F)	21.0 4.31	29.7 5.39	12.2 3.2	27.6 5.18	34.7 7.37	60.54 8.84	52.4 8.98
Flash point, K (°F) Pour point, K (°F)	516(470) 200(-100)	539(510) 217(-70)	489(420) ---	544(520) 211(-80)	519(475) 214(-75)	519(475) 228(-49)	561(550) 213(-76)
Specific gravity at 289 K (60 °F)	1.00	1.00	--	0.995	0.947	0.96	0.986
Total acid number (tan) Mg Koh/g oil	0.07	0.03	0.15	0.40	0.06	0.00	1.01
EHL film thickness h mm (min) L ratio (h/σ)	0.43(17) 0.75	0.52(20) 0.90	0.34(13) 0.58	0.50(20) 0.87	0.66(26) 1.15	0.76(30) 1.33	0.76(30) 1.33
Specification	none b. stock	Mil-L- 23699	Mil-L- 7808J	DOD-L- 85734	DERD- 2487	none	none

load was then increased to 5784 N/cm (3305 lb/in), which gave a pitch-line maximum Hertz stress of 1.71 GPa (248 ksi) if plain bending is assumed. However, because there was an offset load, an additional stress was imposed on the tooth bending stress. Combining the bending and torsional moments gave a maximum stress of 0.26 GPa (37 ksi). This bending stress does not include the effects of tip relief, which would also increase the bending stress.

Operating the test gears at 10 000 rpm gave a pitch-line velocity of 46.55 m/sec (9163 ft/min). Lubricant was supplied to the inlet mesh at 800 cm³/min (0.21 gpm) at 321K (120 °F). The lubricant outlet temperature was nearly constant at 350 K (170 °F). The tests ran continuously (24 hr/day) until they were automatically shut down by the vibration detection transducer, located on the gearbox adjacent to the test gears. The lubricant circulated through a 5-μm fiberglass filter to remove wear particles. After each test the lubricant and the filter element were discarded. Inlet and outlet oil temperatures were continuously recorded on a strip-chart recorder.

The pitch-line elastohydrodynamic (EHL) film thickness was calculated by the method of reference 8. It was assumed, for this film thickness calculation, that the gear temperature at the pitch line was equal to the outlet oil temperature, even though the inlet oil temperature was considerably lower. It is possible that the gear surface temperature was even higher than the outlet oil temperature, especially at the end points of sliding contact. The EHL film thickness and the initial ratio of film thickness to composite surface roughness Λ , for these conditions was computed at the 1.71 GPa (248 ksi) pitch-line maximum Hertz stress and the values are shown in Table 4.

Results and Discussion

Gear Life

The surface pitting fatigue lives of the AISI 9310 gears run with all seven lubricants are shown in figure 3 and Table 5. These data are shown on Weibull coordinates and

were analyzed by the method of reference 9. The life shown is the life of gear pairs in millions of stress cycles or millions of revolutions. A failure is defined as one or more spalls covering more than 50 percent of the width of the Hertzian contact. Typical fatigue spalls for each lubricant along with cross sections of the fatigue spalls are shown in figure 4.

Lubricant A (fig. 3(a)) is the reference oil for these lubricant tests. The 10- and 50-percent system (two gears) lives (the life at 90- and 50-percent probability of survival) were 5.1 million and 20.4 million revolutions or stress cycles, respectively. The failure index was 30 out of 30 which is the number of failures out of the number of tests conducted. A typical fatigue spall with lubricant A is shown in figure 4(a).

The surface pitting fatigue lives of the AISI 9310 gears run with lubricant B are shown in figure 3(b). A typical fatigue spall for lubricant B is shown in figure 4(b). Lubricant B is a 5 cSt lubricant meeting the MIL-L-23699 specification. The 10- and 50-percent system lives of the gears run with lubricant B were 12.1 million and 75 million revolutions or stress cycles, respectively. The failure index for this lubricant was 20 out of 20. These data indicate that the fatigue life of 9310 gears run with lubricant B is approximately 2.4 times that for lubricant A. The confidence number for the life difference between lubricant B and lubricant A was 84 percent which indicates that the difference is statistically significant. The confidence number indicates the percentage of time the order of the test results will be the same. For a confidence number of 84 percent, 84 out of 100 times the test is repeated the gear life with lubricant B will be higher than lubricant A. Experience has shown that a confidence number of 80 percent or greater would indicate a meaningful life difference.

The life difference between lubricant A and lubricant B of over two to one would not be expected based on the small difference in lubricant viscosity and specific film thickness. However, when it is considered that lubricant A does not have an additive package that would include an EP additive,

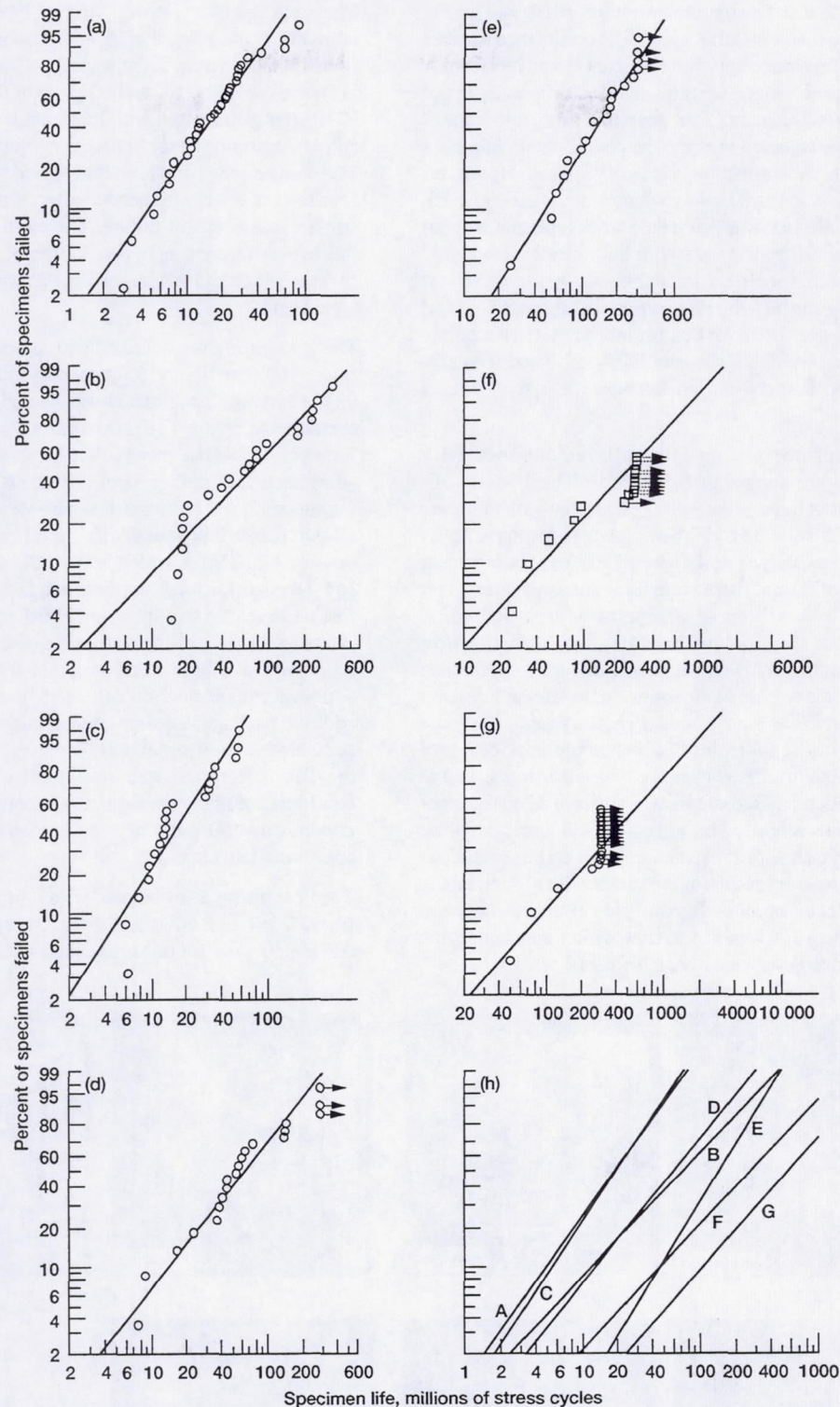


Figure 3.—Surface pitting fatigue lives of AISI 9310 gears run with seven different lubricants. Pitch diameter 8.39 cm (3.5 in.); speed 10 000 rpm; maximum Hertz stress 1.71 GPa (248 ksi); gear temperature 350 K (170 °F). (a) Lubricant A. (b) Lubricant B. (c) Lubricant C. (d) Lubricant D. (e) Lubricant E. (f) Lubricant F. (g) Lubricant G. (h) Summary.

then the life difference is more in line with expected results based on the test conducted in references 3 and 4.

The surface pitting fatigue lives obtained with lubricant C are shown in figure 3(e). Lubricant C is a 3 cSt lubricant meeting the MIL-L-7808J specification. A typical fatigue

spall for lubricant C is shown in figure 4(c). The 10- and 50-percent systems lives of the 9310 gears run with lubricant C were 5.67 million and 20.7 million revolutions or stress cycle, respectively. The failure index for this lubricant was 20 out of 20. These data indicate that the

fatigue life of 9310 gears run with lubricant C was nearly equivalent to that with lubricant A. The confidence number for the life difference between lubricant C and lubricant A was 55 percent which indicates that the difference is not statistically significant. The gear life with lubricant C would not be expected to equal the gear life with lubricant A based on the lubricant viscosity alone. However, lubricant C is a formulation which contained some EP additives while lubricant A is a base stock lubricant without EP additives. Since the tests with both lubricant A and C were run with specific film thickness in the mixed or boundary regime then the EP additives in lubricant C would improve the gear life over that for lubricant A. This points out the need for EP additives in lubricants used for gears operating with specific film thickness less than one as demonstrated in other tests.

The gear surface pitting fatigue lives obtained with lubricant D are shown in figure 3(d). The 10- and 50-percent system lives of the 9310 gears tested with lubricant D were 11.75 million and 50.8 million revolutions or stress cycles, respectively. The failure index for this lubricant was 17 out of 20 and there were three suspended tests that completed 300 million stress cycles without failure. A typical fatigue spall for lubricant D is shown in figure 4(d). The life for lubricant D was 2.3 times that for the reference lubricant A and was nearly identical to the life for lubricant B. Lubricant D has more boundary additive and nearly the same viscosity as lubricant B which could provide better surface fatigue life. However the two lubricants B and D had nearly identical fatigue lives. Lubricant A, on the other hand, has only slightly less viscosity than lubricant D but does not have an additive package or EP additive which is the most probable reason for the shorter life of lubricant A. The confidence number for the life difference between lubricant D and A was 83 percent which means that the life difference is statistically significant.

The gear surface pitting fatigue lives obtained with lubricant E are plotted on Weibull coordinates shown in figure 3(e). A typical fatigue spall for lubricant E is shown in figure 4(e). The 10- and 50-percent system lives of the 9310 spur gears tested with lubricant E were 46.5 million and 152 million stress cycles or revolutions, respectively. The failure index for lubricant E was 15 out of 19 with four tests that were suspended after 500 hr or 300 million stress cycles without failure. The confidence number for the life difference between lubricant E and A was 99 percent which shows that the life difference is statistically significant.

The gear pitting life obtained with lubricant F is shown in figure 3(f). The 10- and 50-percent system lives of the AISI 9310 gears for this lubricant was 45 and 276 million stress cycles respectively. The failure index for this lubricant was 7 out of 17 which means that there were 10 tests that completed 300 million cycles without failure. A typical fatigue spall for lubricant F is shown in figure 4(f). The 10-percent surface fatigue life for lubricant F was 9 times that for lubricant A which is a 5 cSt lubricant and about equal to that for lubricant E which is a 7.5 cSt lubricant. The main reason that lubricant F did not produce a better gear surface fatigue life than lubricant E appears to be that lubricant F is a base stock lubricant without an anti wear additive package while lubricant E contained a good additive package. The confidence number for lubricant F compared to lubricant A was 99 percent which means that the life difference was statistically significant. The confidence number for lubricant F compared to lubricant E was only 50 percent which means the lives were approximately equal.

The gear pitting life obtained with lubricant G is shown in figure 3(g). The 10- and 50-percent system lives of the AISI 9310 gears for this lubricant was 103 and 568 million

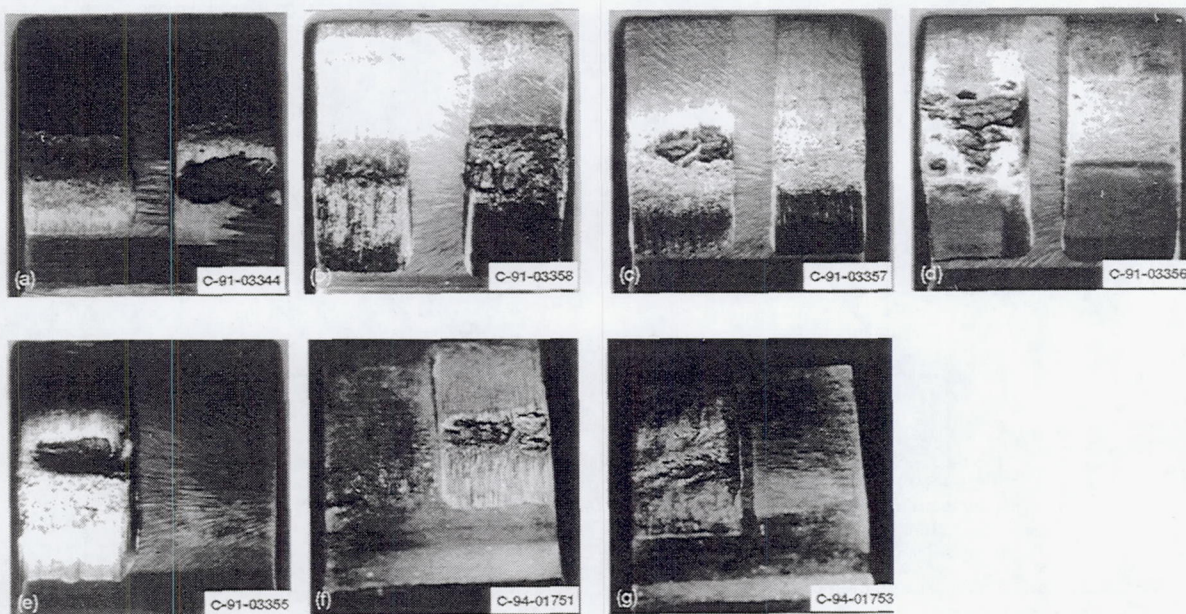


Figure 4.—Typical fatigue spalls for AISI 9310 steel spur gears run with seven different lubricants. Pitch diameter 8.39 cm (3.5 in.); speed 100 rpm; maximum Hertz stress 1.71 GPa (248 ksi); gear temperature 350 K (170 °F). (a) Lubricant A. (b) Lubricant B. (c) Lubricant C. (d) Lubricant D. (e) Lubricant E. (f) Lubricant F. (g) Lubricant G.

Table 5.—Surface Pitting Fatigue Lives of AISI 9310 with Different Lubricants

Lubricant code	Lubricant basestock	Gear system life, millions of stress cycles		Weibull slope	Failure index ^a	Confidence number, ^b percent
		10 percent	50 percent			
A	polyol-ester	5.1	20.4	1.36	30 of 30	—
B	polyol-ester	12.1	76	1.02	20 of 20	84
C	polyol-ester	5.7	20.7	1.46	20 of 20	55
D	polyol-ester	11.8	50.8	1.29	17 of 20	83
E	polyalkylene-glycol	46.5	152	1.59	15 of 19	99
F	polyol-ester	45.2	276	1.04	7 of 17	99
G	polyol-ester	103	568	1.1	5 of 18	99

^a Number of failures out of number of tests.

^b Percent of time that 10 percent life obtained with - each lubricant will have the same relation to the 10 percent life of lubricant NASA A.

stress cycles respectively. The failure index for this lubricant was 5 out of 18 which means that there were 13 tests that completed 300 million cycles without failure. A typical fatigue spall for lubricant G is shown in figure 4(g). The 10-percent surface fatigue life for lubricant G was 20 times that for lubricant A which is a 5 cSt lubricant and was about 2.3 times that for lubricant E which is a 7.5 cSt lubricant and lubricant F which is a 9 cSt lubricant. It appears that the main reason for the life improvement of lubricant G over that for lubricant F which has the same viscosity is the fact that lubricant F has no antiwear additive package while lubricant G has some type of additive that provided boundary lubrication. The confidence number for lubricant G compared to lubricant A was 99 percent which means that the life difference was statistically significant. The confidence number for lubricant G compared to lubricant F was 70 percent which means the life difference was also statistically significant.

These life results are summarized in figure 3(h) and Table 5. The life of lubricant G was more than twenty times that for lubricant A and more than two times that for lubricants E and F. Lubricant G is a 9 cSt lubricant and had a calculated specific film thickness L of 1.33. It was, therefore, expected that lubricant G would produce longer fatigue life than the other less viscous lubricants. However, it could not be analytically determined just how much improvement in surface fatigue could be obtained with this higher viscosity lubricant. The surface fatigue testing was, therefore, necessary to determine gear life with these lubricants.

Figure 5 is a plot of specific film thickness ratio L versus the relative gear surface fatigue life and shows how the gear life is effected by the specific film thickness ratio. The test results are very conclusive in showing that when gears are operated with lubricants that provide specific film thickness around one or greater, the surface fatigue will show large improvements over some of the turbine engine lubricants that provide lower EHL specific film thickness. In addition as the EHL specific film thickness ratio increases above 1 the surface fatigue is further improved.

The above results also point out the need to provide separate lubricants for power transmissions such as turboprop or turbofan reduction gearboxes and helicopter gearboxes that have higher viscosities than the engine lubricants to provide increased life and reliability of these systems.

Summary

Spur gear surface pitting fatigue life tests were conducted with seven lubricants with viscosities from 3 cSt to 9cSt at 372 K (210 °F) using a single lot of consumable-electrode vacuum melted (CVM) AISI 9310 steel spur gears. The gears were case carburized and hardened to a

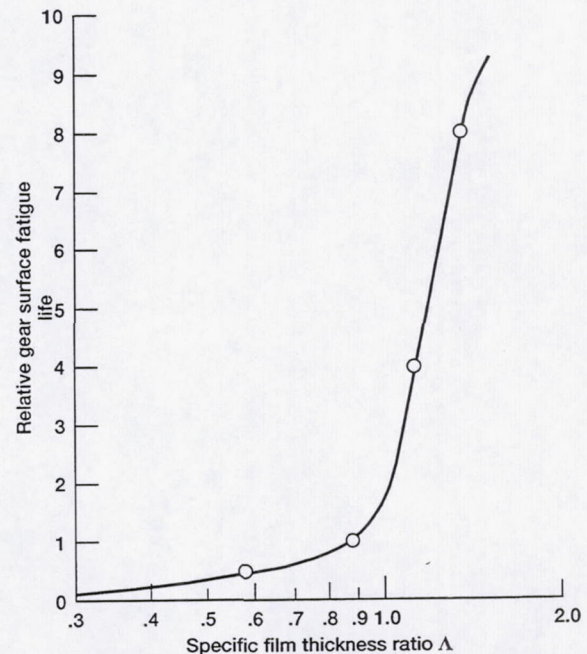


Figure 5.—Relative gear surface fatigue life versus specific film thickness ratio L . Test conditions, 10000 rpm, maximum Hertz stress 1.71 gPa (248 ksi), and a bulk gear temperature of 350 K (170 °F).

Rockwell C-60 and finish ground. The gear pitch diameter was 8.89 cm (3.5 in.). The lot of gears was divided into seven groups, each of which was tested with a different lubricant. The test lubricants can be classified as synthetic polyol-esters with various viscosities and additive packages. Test conditions included a bulk gear temperature of 350 K (170 °F), a maximum Hertz stress of 1.71 GPa (248 ksi) at the pitch line and a speed of 10 000 rpm. The following were obtained:

1. Lubricants with a viscosity that provides a specific film thickness greater than one and with an additive package produced surface fatigue lives that were 4 to 8.6 times that of lubricants with a viscosity that provided specific film thickness less than one.
2. As the lubricant viscosity is increased to give EHL specific film thickness ratios L well above 1 the gear surface fatigue life is further improved.
3. A low viscosity lubricant with an additive package produced gear surface fatigue lives equivalent to a similar base stock lubricant with 30 percent higher viscosity but without an additive package.
4. Lubricants with the same viscosity and similar additive packages gave equivalent gear surface fatigue lives.

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